UTILIZATION OF POWER PLANT WASTE HEAT FOR HEATING

R. W. Timmerman
Consulting Mechanical Engineer
Boston, Massachusetts U.S.A.

ABSTRACT

This paper describes progress to date in development of a new system for the utilization of power plant waste heat for heating buildings, which makes power plant waste heat competitive with heavy oil, and possibly cheaper than light oil under present day New England conditions. This system consits of modifications to the power plant condenser to produce water at temperatures between 105F and 120F, a novel once-thru piping scheme, and a multistage heat pump and control system. The heated water is carried underground in ordinary reinforced concrete pressure pipe. The paper presents results of preliminary economic studies which show that waste heat appears to be an economically viable source of heating energy at the present stage of technology. The paper concludes with a discussion of further research needs.

INTRODUCTION

In recent years there has been some interest in the utilization of power plant waste heat. In addition to uses such as heating greenhouses, and aquaculture, use of waste heat for heating buildings has been investigated. [1,2]

My own interest dates from the early 1970's, but I did not begin the work on which this paper is based until early in 1975. This paper is in the nature of a progress report of work to date, and as such does not represent finished work. Please note also that much of the technology described in this report is covered by U.S. Patents Pending.

Two schemes for the use of power plant waste heat for heating buildings have evolved: High temperature schemes employing extraction steam, and low temperature schemes which use waste heat directly as it comes from the condensers. The various cogeneration schemes are a subset of the high temperature schemes. This paper presents a third alternative, which is a medium temperature scheme, operating at supply temperatures on the order of 105 to 120F.

The remainder of the paper is divided into 3 parts: definition of the problem, the solution proposed, and the results of the economic study.

DEFINITION OF THE PROBLEM

Simply stated, the problem of waste heat utilization is to find economical uses for power plant waste heat. In the case of heating systems, it is possible to develop a set of engineering guidelines, which serve to further define the problem, and also point in the direction of a possible solution.

Engineering Guidelines

1. The scheme should require no major power plant modifications, nor major design changes in new plants. This makes it possible to use the waste heat output of existing plants, and keeps the capital costs down. It also reduces the amount of new technology that must be sold to the generally conservative utility industry.

2. The cost of interference with normal power plant economic dispatch is to be borne by the heating customers. For example, if the plant must run to meet the heating load, at a time when it is not economical for power generation, the cost of operation will be borne by the heating customers. Likewise, the cost of any degradation of the heat rate is for the account of the heating customers.

3. Total cost of heat delivered in a form the customer can use should be competitive with oil or gas, also in that form. This generally means charging boiler efficiency against oil or gas.

4. Likewise, the economics should be based on real world conditions. The carrying charges should include realistic taxes and interest rates. The load factor should be a realistically achievable value, without resort to unrealistically high summer loads.

5. Sufficient standby capacity should be provided to cover normal operating contingencies, such as power plant outages. Murphy's Law applies to power plants, so it is not realistic to assume that all maintenance can and will be done in the off season.

6. The heat must be delivered in a form that can be used by at least the majority of existing building heating systems.

These guidelines seem to be a bit strict, but I believe that they are necessary. Waste heat systems are capital intensive, and unless they are government subsidized, the capital will have to come from the normal capital markets. The only way of convincing investors to put up the money is to convince them that they stand a reasonable chance of getting their money back, with interest.

Implications of Engineering Guidelines

Although most of the implications of these guidelines are fairly straightforward, there are some that are rather subtle, but with far-reaching implications.

One inexpensive way to meet the standby requirement is to use a grid connected system, supplied by several heating sources. This offers the same benefit to heating systems that power grids have, namely reduction in the amount of standby capacity. However, in order to feed several sources into a grid, they all must be compatible. If the heating medium is hot water, all sources must discharge at about the same temperature. Generally it is less expensive to produce a stream of fluid at temperature T directly, than it is to produce it by mixing two streams, one at T minus ten degrees, and the other at T plus ten degrees, due to the irreversible nature of the mixing process. In addition, suddenly introducing a stream of fluid that differs sufficiently from the average to change the average temperature will probably create some problems for the terminal apparatus.

Neither schemes based on extraction steam, nor schemes based on condenser water can meet this criterion. Both turbine extraction pressures and circulating water outlet temperatures vary with load on the machine. This is a more severe problem for systems using extraction steam, because the extraction steam pressure is not only the determinant of energy level in the steam, but is also the driving force producing flow in the piping system. Therefore, if the pressures at the various plants deviated very far from the equilibrium value, the flow pattern would not be what was desired. Systems using circulating water would only be troubled by problems of temperature incompatibility, which are more manageable. The need to meet electric loads in a manner that would not cause great temperature and pressure incompatibility problems for the heating system is an extremely difficult and expensive dispatching problem.

The requirement that deviations from normal dispatch be borne by the heating system discourages systems that require certain plants to be run to meet heating load, regardless of power load. Likewise, systems that provide power as a by-product of heating are at an economic disadvantage, because the power generated is not firm power, but varies with heating load. Since standby facilities are needed to meet the power load when there is no heating load, the power produced as by-product is priced as by-product power, instead of firm power. This reduces the credit for power production.

DESCRIPTION OF THE UTILIZATION SYSTEM

My scheme is an attempt to avoid these problems and still meet the economic requirements. I propose to use a grid-connected system to insure reliability. The heat source will be condenser circulating water at a controlled temperature of 100F to 120F. Use of a controlled outlet temperature at this level has two benefits: The water temperature is closer to the required utlization temperature, and all plants are made compatible with each other. The cost of the heat will be determined by the difference in heat rate of the various power plants under normal conditions and the heat rate while delivering water at the controlled outlet temperature. This particular choice of outlet temperature comes about from trying for the highest possible outlet temperature that existing turbines are capable of. This avoids the problem of selling power companies on new technology, and avoids building special purpose plants which have poor performance when heating is not required. The back pressure of existing turbines is limited to 5 inches of mercury, which works out to 133.75F, which in turn makes possible a maximum outlet temperature in the 100 to 120F range.

Conduit Design

A great simplification of the conduit design is possible in the 100 to 120 F temperature range. Most underground heat conduits have a well deserved bad reputation for high installed costs, and for high maintenance costs. However, ordinary reinforced concrete pipe is permitted by the piping code for temperatures under 150F. Heat loss of large diameter pipes is on the order of one or two degrees in ten miles, with ordinary sand used as insulation. Sand has a number of advantages over other insulation materials: It is readily available at low cost, installation is simple and fast, it drains water easily, and it is not damaged by water. The last benefit is rather important, because underground conduits will flood, and therefore should be designed so as not to be damaged by flooding.

Ordinary bell and spigot pipe joints have sufficient flexibility to accommodate thermal expansion. Guides are not needed, because the sand insulation has sufficient bearing capacity to support and guide the pope. This obviates the need for manholes to contain the exapnsion joints and guides, reducing both the construction cost of the manholes, and the maintenance cost of the expansion joints. As with any bell and spigot joint system, anchors are needed at changes of direction. However, by elimination the expansion joints, the anchors can be simpler, cheaper and farther apart.

Supply and Return Piping Configuration

Two configurations of supply and return piping are possible: conventional two pipe design and a once-thru system. The conventional two

pipe system employs supply and return lines, with fluid flowing in a circuit. This is the conventional approach, and is most economical for short and intermediate range transmission.

In addition, it is possible to dispense with the return line if both the plant and the utilization point are located on a common body of water. In this case the water is taken in at the power plant, heated in the power plant, and transmitted to the point of use. It is cooled at the point of use, and discharged back to the body of water.

The once thru system has an inherently low utilization efficiency, since the plant takes in water at temperatures in the 32 to 40F range in the winter, and discharges at temperatures in the 100 to 120F range. Utilization apparatus will probably not be able to cool the water down past, say 80F. This results in utilization efficiencies on the order of 50%.

For a given power plant output, a one pipe system will have roughly half the water flow of a two pipe system, since the utilization apparatus operates through about the same temperature difference in both cases. The once thru system will use only half the plant output, but to do this will move only half the water, thru half the distance, resulting in a saving in both capital cost and pumping power cost. As a result, the best application of a once thru system is for long distance transmission of heat from a plant with a cheap heat source, such as coal or nuclear.

Conversion Systems at the Point of Use

There are three obstacles to the direct use of waste heat in building heating systems: low supply temperatures, high temperature drop, and possible corrosion problems.

The supply temperature in the 120F range is slightly lower than commonly used supply temperatures in building heating systems. Furthermore, a fairly high temperature drop between supply and return is essential to the economics of the system, by keeping transmission and distribution pipe sizes within reason, and by keeping the utilization efficiency up on once thru systems. High temperature drop implies a low outlet temperature, probably in the 70 to 90F range. This results in a much lower average temperature than most building heating systems are designed for. While it should be possible to design a heating system for a new building for these conditions, it probably is not practical to retrofit an existing heating system for these temperature levels.

A third problem is possible corrosion. Many of the systems along the coast will employ salt water. Even if salt water is not used directly in a two pipe, closed loop system, there is still the possibility of salt entering the system as plants are switched in and out of the system. In order to protect the building heating system from corrosion due to salt water, it becomes necessary to install some sort of heat exchanger

to isolate the waste heat system from the building heating system.

Instead of a passive heat exchanger, it is possible to use a heat pump. This will isolate not only the salt water, but the low temperature levels as well. A heat pump will make it possible to operate the building heating system at some higher temperature level than the waste heat system, thereby reducing the retrofit problems.

ENGINEERING DETAILS

While the system has been outlined in broad detail, there are two crucial parts of the system which I would like to discuss in greater detail. These are the power plant modifications necessary to produce 120F water, and the terminal apparatus.

Power Plant Modifications

One of the goals of a good waste heat system is minimum interference with normal power plant operation and design, and especially, minimum interference with normal economic dispatch. Since the heating and power loads vary independently of each other, it follows that some means must be provided to decouple the two.

The grid connected system permits plants to be added or subtracted from the heating system as needed to meet the heating load. It is also feasible to introduce some storage into the system to meet short term peak loads. In order to be able to switch a plant in and out of the heating system at will, it is necessary that the plant modifications for producing 120F water do not interfere with normal plant operations. In particular, the modifications must not affect the heat rate of the plant when it is not delivering heat to the system.

Conventional power plant condensers are designed for water temperature rises on the order of 20 to 40F. With wintertime water temperatures in the 35 to 40F range, the outlet temperatures will fall in a range between 55F and 80F, which is well below the desired 100 to 120F range. If the plant is part of a two pipe system, the temperature rise needed will be on the order of 40 to 60F, which is at the high end of the normal operating range. Therefore, some means must be provided to produce tne desired temperature rise, while keeping the condensing temperature as close as possible to the outlet water temperature, and not restricting the output of the turbine.

The approach temperature is the difference between the condensing temperature and the circulating water outlet temperature. A circulating water outlet temperature on the order to 120F is desired, without exceeding the maximum condensing temperature of 133.75F (5 inches of mercury back pressure). This allows for an approach of 13.75F, instead

of the normal design approach of 20 to 30F. Refer to Figure 1, which shows the temperature conditions inside a condenser.

In the following discussion, the notation below will be used:

T_{cond}	Condensing Temperature
T_{cwin}	Circulating Water Inlet Temperature
T_{cwout}	Circulating Water Outlet Temperature
R	$T_{cwout}-T_{cwin}$
E	Condenser Effectiveness
U	Overall Heat Transfer Coefficient
A	Surface Area of Condenser
C	Hourly Heat Capacity of Cooling Water (Cooling water mass flow times specific heat)

By definition,

$$E = \frac{T_{cwout}-T_{cwin}}{T_{cond}-T_{cwin}} = \frac{R}{T_{cond}-T_{cwin}} \tag{1}$$

This can be rearranged to read:

$$T_{cond}-T_{cwout} = \frac{(1-E)}{E} R \tag{2}$$

Waste heat applications usually require R on the order of 2 to 4 times design, with an approach ($T_{cond}-T_{cwout}$) on the order of half design. Examination of equation 2 shows that E must be increased by a considerable amount in order to produce the desired approach.

For a condenser:

$$E = 1- \exp(-UA/C) \tag{3}$$

With existing condensers, A is fixed. With the condenser for a power plant being designed, A can be varied, but economic limitations will place an upper limit on A. Examination of equation (3) shows that as C decreases, E increases. This is partially offset by the reduction in U which takes place when the velocity thru the tubes is reduced, which in turn is a direct consequence of reducing C. If a condenser originally

designed for single pass design is converted to multipass design, the mass flow thru the condenser can be reduced, without reducing the velocity, thereby keeping U at or near the design values. At the same time, the reduction in C will permit an increase in E, the effectiveness.

Although research in this area is still underway, preliminary results are that a single pass condenser designed for 25F water temperature rise and 32F approach can be used successfully to produce water at 112F when converted to multipass.

Another reason for keeping the water velocity in the tubes at or near design values is to reduce corrosion problems in the tubes. It is well known that certain minimum water velocities are needed to preclude condenser tube fouling.

Under certain conditions, it may be possible to operate a multipass condenser with multiple pressure zones. When this can be done it reduces the heat rate penalty which is incurred when discharging heat at 120F. The heat rate is improved because the average back pressure is reduced in multipass operation over single pass operation.

While space does not permit a discussion of the technique for switching from single space to multipass, I would like to point out that a technique has been developed which permits switching to take place without interrupting normal circulating water flow. This should permit such conversions to occur with the power plant on line.

Heat Pump

The need for the heat pump has already been discussed. While previous work in waste heat utilization has used single stage heat pumps [1,2], there are some inherent limitations of single stage heat pumps that render them not particularly well suited for waste heat utilization.

Figure 2 is a T-S diagram of a conventional Carnot cycle heat pump. The heating and cooling which the heat source and heat sink undergo are shown by the dotted lines. The Carnot cycle requires constant temperature heat addition and rejection. In this case, heat addition and rejection do not take place at constant temperature. Consequently, there is a loss in efficiency, due to the irreversible nature of heat transfer thru a finite temperature difference. While the Brayton cycle would be a much better match to the conditions, the heat transfer between the gas used as a Brayton cycle working fluid and the liquid used to carry waste heat is inherently poor, negating much of the advantage.

Another approach is to use a series of Carnot cycles to approximate the actual cooling and heating curves. This is illustrated in Fig. 3. A comparison has been made between a single stage heat pump and a 3 stage heat pump, both with the same heat exchanger terminal differences and the same compressor efficiency. Both heat pumps take in water at 120F,

discharge it at 90F, while heating water from 130F to 160F. Under the conditions used in the study, the single stage heat pump had a COP of 5.32, while the 3 stage machine had a COP of 6.95. The 3 stage machine will require 78% of the power input of the single stage machine.

The second problem is the difficulty in economically matching the output of the heat pump with the load. The output of a convector is governed by equation 4: [3]

$$H = c(T_s - T_a)^n$$

H	Heating output
c	Rating constant, determined by test
T_s	Average supply temperature
T_a	Air temperature
n	1.3 for cast iron radiators 1.4 for baseboard radiation 1.5 for convectors

For example, if a system of baseboard radiation was designed for 160F supply temperature, and 130F return temperature (average temperature 145F), with air temperature at 65F, reducing the output to half would require an inlet temperature of 121F and a return temperature of 106F, assuming constant mass flow thru the baseboard unit.

An all air system can be operated with either constant air volume, and variable supply temperature, or constant supply temperature and variable air volume. The later approach has become increasingly popular in recent years. It is not the approach of choice for waste heat utilization.

The least expensive way to operate a waste heat utilization system is to control the temperature of a heated space by varying the supply temperature of the fluid heating the space. This reduces the heat pump temperature lift at low loads. Lower temperature lift translates into higher COP, and less power consumption. This is illustrated by Figure 4, which shows the frequency of occurance of outside air temperatures for Boston, and also the hot water supply temperatures required to meet the heating loads created by these outside temperatures.

Heat pumps designed for waste heat applications will deliver only a fraction of the heating output when used for air conditioning. This is because the compressor suction pressure in cooling applications is only a fraction of the suction pressure in heating applications. The low suction pressure reduces the inlet density, reducint the compressor capacity. I estimate that the cooling capacity of a heat pump designed for waste heat utilization is only 20% of the heating capacity.

ECONOMICS

In order to accurately assess the economics of the system, I have made what I believe are rather conservative assumptions. I assumed that the waste heat transmission and distribution system would be owned and operated by an electric power company, which would have a fixed charge rate of 20%. I assumed that the terminal equipment would be owned and operated by the customer, with the same fixed charge rate. Electric power costs of 5¢ per kwhr and oil costs for steam power plants of $2.00 per million btu were used.

Transmission and distribution piping costs were computed from various pipeline costs reported in Engineering News-Record, and are based on construction in moderately congested areas (not heavily congested downtown areas).

Owning and operating costs of the heat pump were based on the following:

Fixed charge rate of 20%
Cost of heat pump @ $225/ton of capacity (15000 btu/hr heating)
Credit for cooling capacity of 20% of heating capacity
Carrying charges applied to cost of heat pump, less 20% cooling credit, less cost of a heating boiler of the same capacity.
Yearly average COP of 14
Electric power cost of 5¢/kwhr
Diesel fuel at $3.00/million btu

For comparison purposes, heating costs using fossil fuel were computed on the following basis:

Boiler efficiency 80%
#2 Fuel oil at 47¢/gallon ($3.40/million btu.)
#6 Fuel oil at $2.00/million btu
No carrying costs for the heating boiler are included, since the carrying costs for the heat pump are computed on a differential basis with a conventional heating boiler.

Two cases were considered. The first is a residential district, which is principally 5 storey brick row houses. In this case, electricly driven heat pumps would be used. The second case is a large institution, using Diesel engine driven heat pumps with recovery of the exhaust heat, and heat from the water jacket. Both cases are described in Table 1, and the economics are compared in Table 2.

Both of these cases are based on a one pipe system transmitting heat from a fossil fueled power plant. Two more cases investigated differences in cost of a two pipe system, and differences in cost of heat from a nuclear powerplant. Bost of these studies were concerned with costs up to the distribution system only.

Case 3 is a comparison of the cost of heat delivered to the distribution system from a two pipe transmission system supplied by a fossil plant. The conclusions of the analysis are that a two pipe system is slightly less expensive than a single pipe system for the distribution of the heat from oil fired plants due to the higher utilization efficiency. Case 3 is presented in Table 3.

Case 4 is an investigation of the cost of heat from a plant burning cheap fuel, located some distance from a center of utilization. These costs are typical of the economics of waste heat from nuclear plants. Cost of waste heat from base load coal fired plants will be only slightly higher. Case 4 is presented in Table 4.

In order to establish a point of reference for these fuel costs, it is necessary to compute the cost of heating by oil. With the base figures given above, the cost of heating by #2 fuel oil is $4.25/million btu, and the cost of heating by #6 oil is $2.50/million btu.

We can draw some conclusions from these results:

1. Waste heat from fossil fueled power plants is cheaper than #2 fuel oil, up to 10 miles from the plant, under present New England Conditions.

2. Waste heat is about 15% more expensive than a large user than #6 fuel oil, under the same conditions as 1 above.

3. Waste heat from a nuclear plant is cheaper than waste heat from a fossil fuel plant, up to at least 30 miles from the plant.

RESEARCH NEEDS

This paper represents work in progress, and does not represent final conclusions. A considerable amount of further work needs be done before the economic feasibility of power plant waste heat for heating is established. I have mapped out a research program to further investigate the technical and economic feasibility of waste heat utilization. In the order in which they should be done, these steps are:

1. Investigate the suitability of actual power plants as heat sources.

2. Design, build and test a prototype heat pump. Compute cost to manufacture and sell.

3. Design, install, and test a prototype underground heat pipeline. Estimate cost to install in large quantities.

4. Select an area for a demonstration project. Layout the system, and estimate the cost. Model several years operation of the system, and determine savings over oil.

5. From 4, determine economic feasibility. Redesign if necessary and possible.

6. Build and test an actual demonstration project.

The intriguing thing about use of power plant waste heat for heating is that it offers the potential of economically saving large quantities of energy with relatively low development costs.

REFERENCES

1. Aamot, Haldor W. C., Management of Power Plant Waste Heat in Cold Regions, NTIS REPORT AD/A-003 217 December 1974

2. Ileri, A., Reistad, G. M., and Schmisseur, W. E. Urban Utilization of Waste Energy From Therman-Electric Power Plants, ASME paper 75-Pwr-12

3. ASHRAE Guide, 1967, pp 348-349

TABLE 1

DESIGN BASIS OF COST STUDY

ITEM	CASE 1	CASE 2
Transmission System	10 Miles: Plant to Distribution 3 Miles: Distribution to body of water	10 Miles: Plant to Distribution 3 Miles: Distribution to body of water
Distribution System	Small lines serving each building	Few large lines to central heat pumps
Heat Pump Drive	Electric	Diesel
Type of buildings served	5 storey brick row house	Large instutional
Utilization Efficiency	37.5%	50%

TABLE 2

COST COMPARISON OF CASES 1 AND 2

ITEM	CASE 1	CASE 2
Cost of heat at plant	$0.85/million btu	$0.64/million btu
Transmission and pumping	$0.75/million btu	$0.75/million btu
Distribution	$0.31/million btu	$0.16/million btu
Heat pump carrying cost	$0.40/million btu	$0.50/million btu
Heat pump operating cost	$1.05/million btu	$0.80/million btu
Total cost	$3.36/million btu	$2.85/million btu
Cost of heating with #2 oil	$4.25/million btu	
Cost of heating with #6 oil		$2.50/million btu

TABLE 3

CASE 3 COST COMPARISON

ITEM	1 PIPE	2 PIPE
Line length supply and return	13 miles	20 miles
Transmission cost	$0.75/million btu	$1.15/million btu
Efficiency of use	37.5%	100%
Cost of heat at 37.5% efficiency	$0.85/million btu	
Cost of heat at 100% efficiency		$0.32/million btu
Total Cost	$1.60/million btu	$1.47/million btu

TABLE 4
CALCULATION OF COST OF NUCLEAR HEAT

Design Basis

Hours of operation	4800 hrs/yr
Plant size	1100 Mw
Heat produced per kwhr generated	8500 btu/kwhr
Fuel cost	\$0.30/million btu
Utilization efficiency	37.5%

Pipe Data

Size	96 inch
Pressure drop/100 ft.	0.228 feet of water @ 240,000 gpm
Pipe cost, installed	\$300.00/foot

Calculations

Heat moved: 4800 hrs/yr X 8500 btu/kwhr X 1,100,000 kwhr/hr X .375 = 16.83 X 10^{12} btu/yr

Pumping Power: $$\frac{\text{840,000 gpm} \times \text{8.31 lbs/gallon} \times \text{0.228 ft./100 ft.} \times \text{52.8 100 ft./mile} \times .746}{\text{33000 ft. - lbs/hp} \times \text{0.80 (Pump effic.)} \times \text{0.90 (Motor effic)}}$$

= 754 kw/mile

Total yearly pumping cost — \$181,000/mile - year

Carrying cost per mile: 300 dollars/ft. X 5280 ft./mile X 0.20 fixed charge rate = \$181,000
(Per year)

Total cost per million btu/mile: (\$350,000 + \$181,000)/16.83 X 10^6 = \$0.0316

Cost of moving 30 miles	\$1.00/million btu
Cost of heat at plant @ \$0.30/million btu	\$0.10/million btu
Total cost for heat and 30 mile transmission	\$1.10/million btu

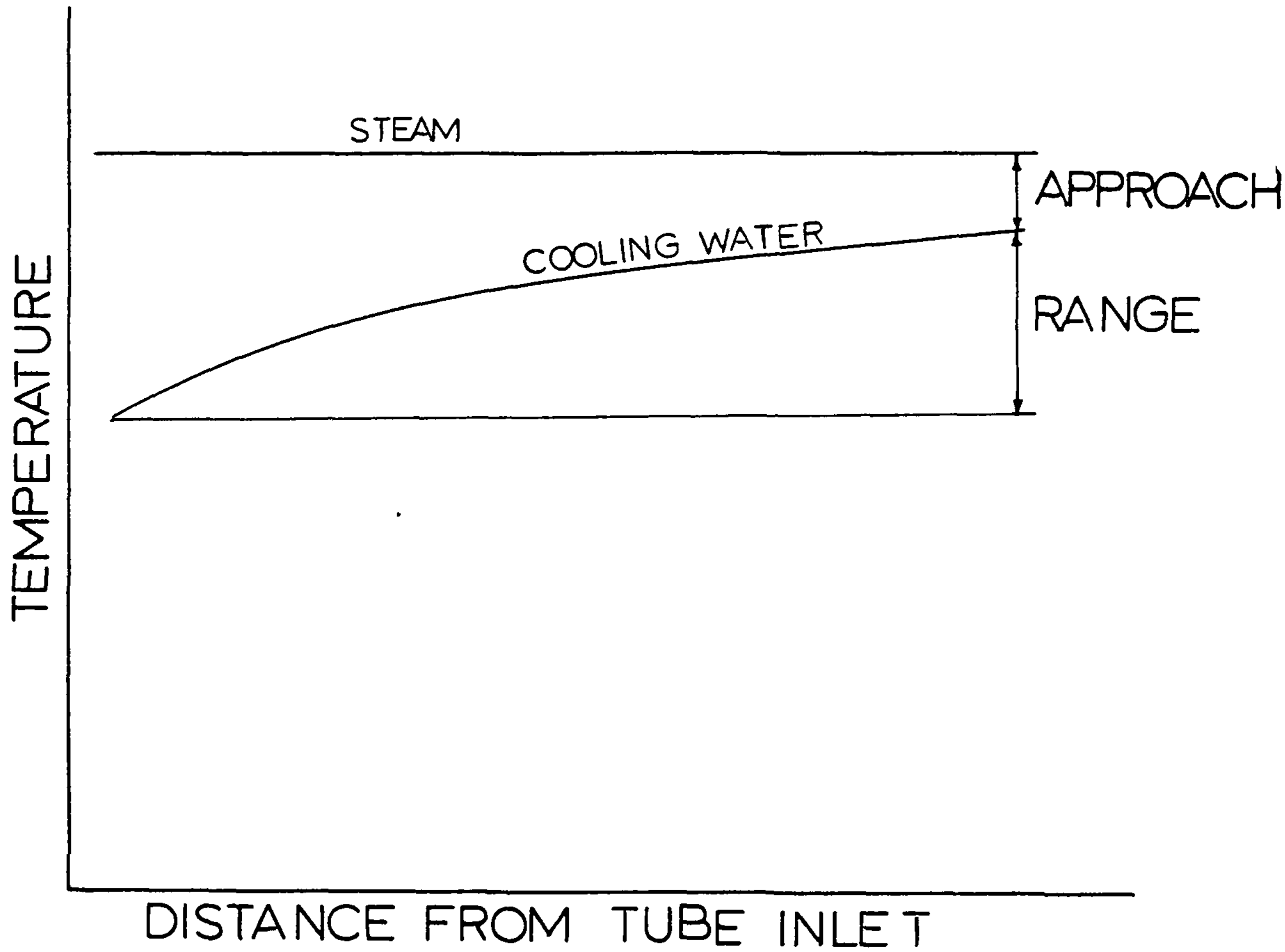

FIGURE 1

TEMPERATURE PROFILE IN A CONDENSER

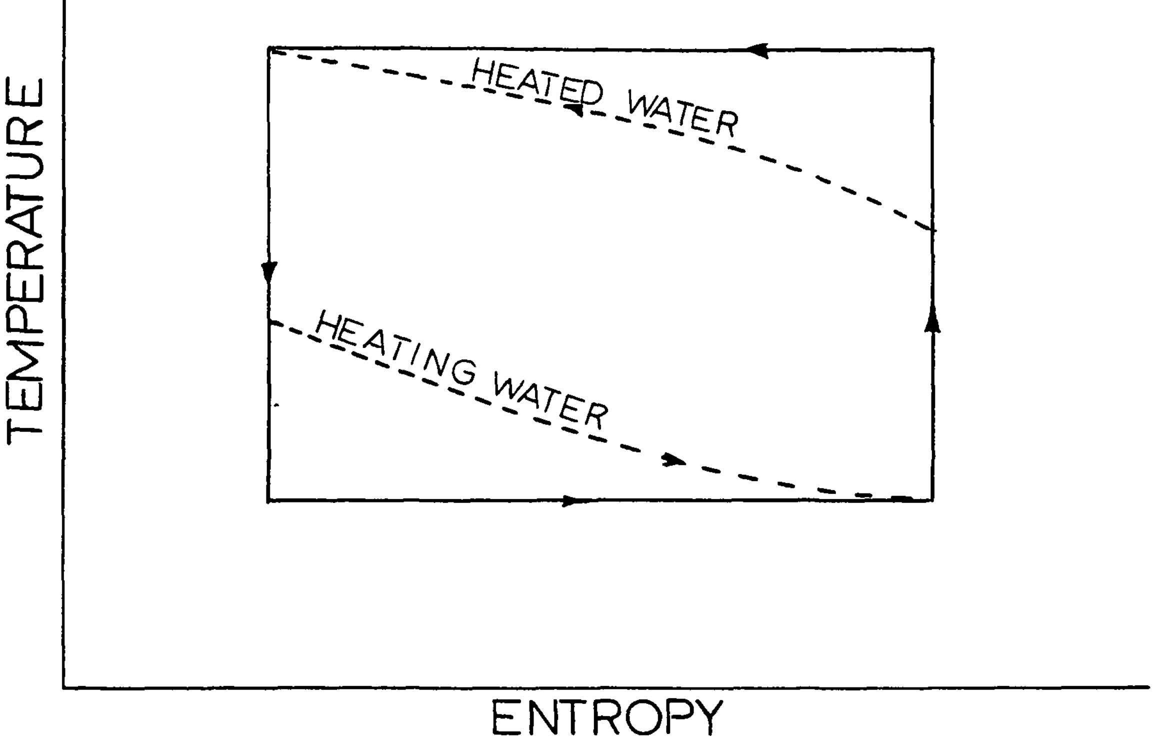

FIGURE 2
T-S DIAGRAM FOR CARNOT CYCLE HEAT PUMP

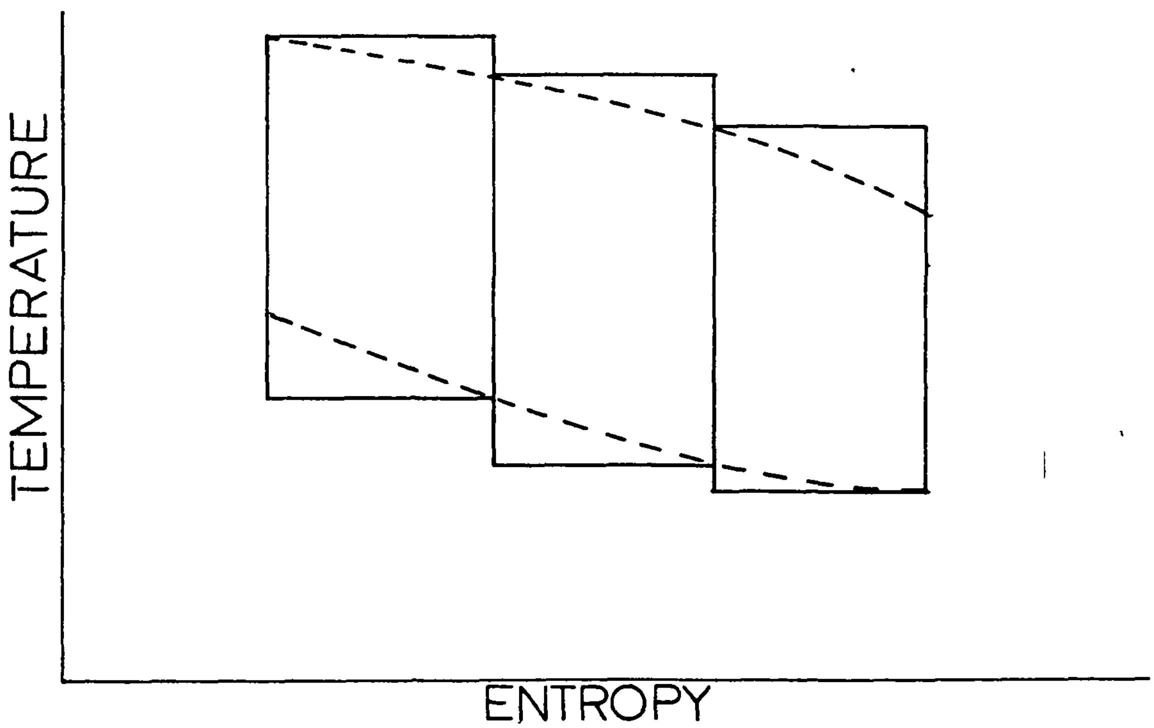

FIGURE 3
T-S DIAGRAM FOR MULTISTAGE HEAT PUMP

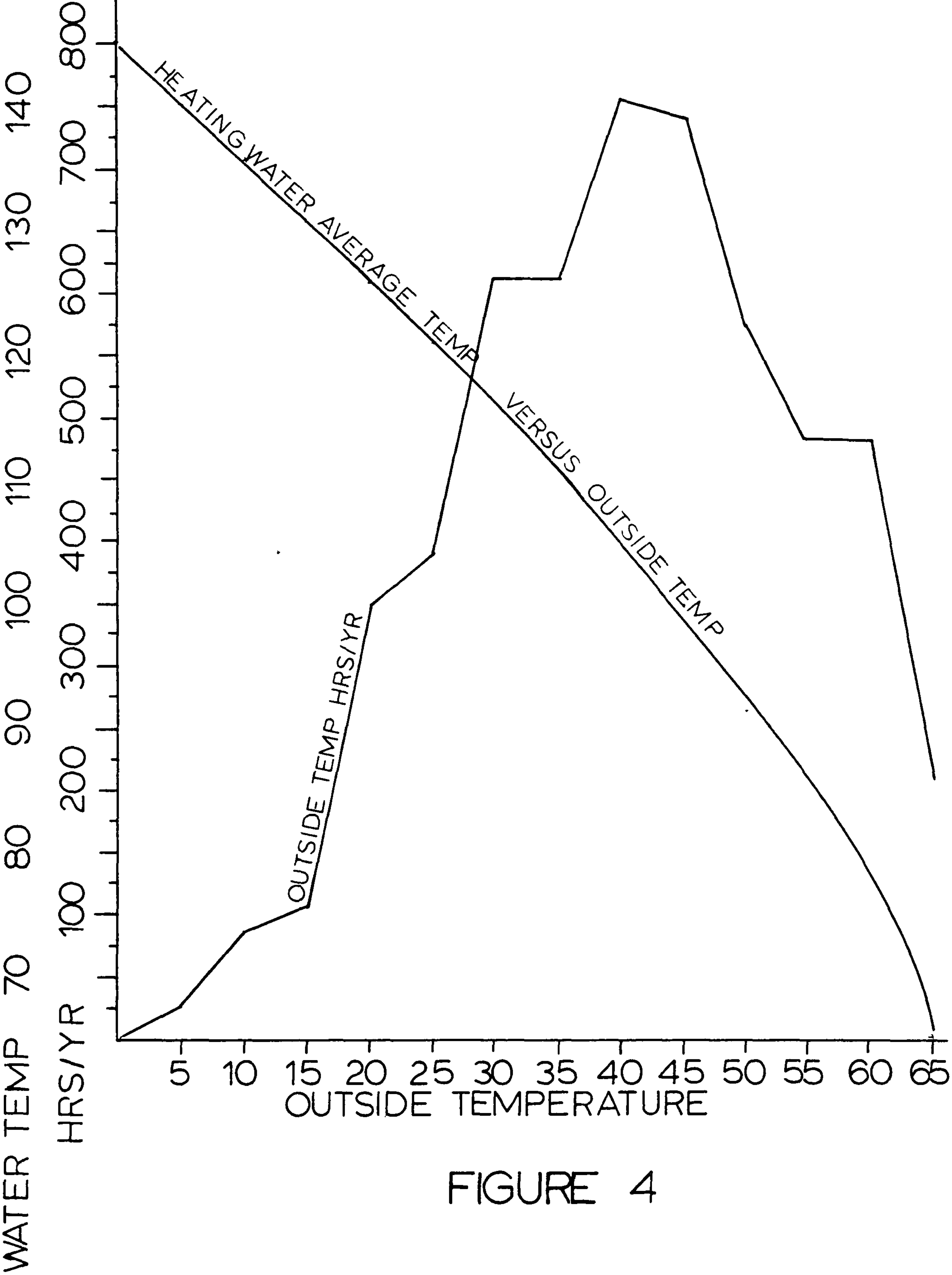
HEATING WATER AVERAGE TEMP VERSUS OUTSIDE TEMP
OUTSIDE TEMP HRS/YR
WATER TEMP
70 80 90 100 110 120 130 140
HRS/YR
100 200 300 400 500 600 700 800
5 10 15 20 25 30 35 40 45 50 55 60 65
OUTSIDE TEMPERATURE

FIGURE 4